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Mail Stop Non-fee Amendment
Commissioner for Patents
P.O. Box 1450
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Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
File date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 file date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

INTRODUCTORY COMMENTS

This is the 5th amendment to this CIP application no. 10/643274. The 1st amendment was dated 27 NOV 2003. The 2nd amendment was dated 27 FEB 2004. The 3rd amendment was dated 23 MAR 2004. The 4th amendment was dated 30 MAR 2004.

This 5th amendment is to: (1) correct a fatal flaw in the specification that fails to properly describe the intake stroke in the 4-Stroke embodiment of this engine, (2) include a version of the 1-way clutch in the written specification that references the changed FIG 12 on the last sheet and (3) revise the **Underlying Mathematics** subsection to make it easier to understand by reducing 5 examples to 1 and more comprehensive with additional equations.

See the REMARKS on sheet 13.

This amendment is believed to be in agreement with Revised Amendment Practice – Effective Date: July 30, 2003. Hopefully, it will be the last for this application.

I can be contacted by the above email or telephone. If by phone, the best time to call is 0730-0830 Hawaii time, 6 hours later than East Coast daylight saving time.

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Underlying Mathematics.

Definitions:

1 BTU = 778 ft-lbf
1 hp = 550 ft-lbf/sec.
2πr′ = length of 1-way clutch rim at connecting rod contact. (ft)
bore – cylinder diameter. (in.)
Cp – cylinder pressure calculated from known bore size. (psi)
Dp – displacement (cu.in.)
E – fuel efficiency
F – combustion force per piston. (lbf)
Fg – fuel flow rate (gals/hr)
Fi – force on the inner race (lbf)
Fr – fuel flow rate (lbm/sec)
Fu – force per unit 89 (lbf) See FIGs 7 or FIG 8 for unit 89.
Fw – fuel's weight (lbm/gal.)
hp – shaft horsepower.
k = 2 or 4 (k = 2 for a 2-stroke. k = 4 for a 4-stroke.)
Lo – ~~power losses~~ [[( ]]fraction of power lost ~~fuel's energy density).~~
n – number of active pistons. 2,4,6,8, …
n/k – number of overlapping pistons cycling through the power stroke.
Nu – number of units 89 (FIGs 7,8).
[[F′]] **Pp** – estimated combustion pressure per piston. (psi) Used to find the bore size. (in.)
Ps – length of piston's stroke. (in.)
Qc – fuel's energy density. (BTU/lbm)
r – radius of cylinder. (in)
r′ – 1-way clutch radius at connecting rod contact. (ft)
ri – radius of the 1-way clutch inner race. (ft)
Rv – power shaft's rotation rate. (rpm)
Sp – Center to center spacing between units 89 (FIGs 7,8). (ft)
T – torque per piston. (lbf-ft)
T′ – total shaft torque. (lbf-ft)
Vp – piston velocity. (ft/sec)

Equations:

Vp = π(r′)(Rv)/(30) Piston rod's and the 1-way clutch's rim speeds are equal at contact.
r′ = 30(Vp)/π(Rv) **r′,Vp, Rv** are central to this engine's design and operation.
Rv = 30(Vp)/(πr′)
F = 550hp(k)/(nVp)
~~**F = 16500(hp)(k)/π(n)(Rv)(r′)**~~
hp = F(n)(Vp)/550
hp = Fr[778(Qc)(1-Lo)]/550
T = F(r′)
T′ = nT/k
[[F′]]Pp = F/[π(r²)]

$r^2 = F/\pi[[F']]P_p)$

$bore = 2[F/(\pi[[F']]P_p)]^{.5}$

$F = \pi[[F']](P_p)(bore^2)/4$

~~Sp = 550hp(1+lo)~~

~~Fr = (Sp)/778Qe)~~

~~Fr = (F)(n)(Vp)/[k(778Qe)]~~

$Fi = F(r')/ri$

$Nu = 2\pi(ri)/Sp$

$Fu = F(r')(Sp)/2\pi(ri^2)$

$Fu = F(r')/(ri)(Nu)$

$Fu = Fi/Nu$

$Cp = 4F/(\pi bore^2)$

$Dp = \pi(bore/2)^2(Ps)(n)$

$Fr = 550hp/[778(Qc)(1\text{-}Lo)]$

$Lo = 1 - 550hp/778(Qc)Fr$

$E = 1 - Lo$

$E = 550hp/778(Qc)Fr$

$Fg = Fr(3600)/(Fw)$

The following example demonstrates the effectiveness of the Underlying Mathematics in finding the correct general engine specifications from which the rest of the engine can be built. The given values are hypothetical. This example is for a low power engine, e.g. lawn mowers and outboard marine, but the math can be applied to any size engine.

Example:

Given:

[[F′]]Pp = 100 psi; **F** = 300 lbf; **Vp** = 3.5 ft/sec; **r′** = 4.5" = .375 ft; ri = 3.75" = .3125 ft;
k = 2; **n** = 2; **Qc** = 20500; **Lo** = .35; **Fw** = 6 lbm/gal; Sp = 6" = .5ft

hp = 300(2)(3.5)/[2(550)] = **1.909**

$\mathbf{r^2} = 300/(100\pi) = \mathbf{.9549}\ \text{in}^2$

bore = $2[300/(100\pi)]^{.5}$ = **1.9544** in.

Rv = $30(3.5)/(.375\pi)$ = **89.13** rpm

Fi = 300(4.5)/3.8 = **355.37** lbf

Nu = $2\pi(3.8)/6$ = **4**

Fu = 300(4.5)/[6(3.75)] = **60** lbf.

T = 300(.375) = **112.5** lbf-ft

Fr = 550(1.909)/[778(20500)(1-.35)] = **.000101284** lbm/sec.

Fg = .000101284(3600)/6 = **.060770629** gals/hr.

Given: **hp** = 10; **F** = 380 lbf.

Vp = 550(10)(2)/(2)(380) = **14.5** ft/sec.

Rv = $30(14.5)/(.375\pi)$ = **369** rpm.

Fi = 380(4.5)/3.8 = **450** lbf

Fu = 380(4.5)/[6(3.75)] = **113** lbf.

Cp = 4(380)/[(1.9544^2)π] = **126.67** psi.

T = 380(.375) = **142.5** lbf-ft

Fr = 550(10)/[778(20500)(1-.35)] = **.000530537** lbm/sec.

Fg = .000530537(3600)/6 = **.318322345** gals/hr.

~~The examples next are only to illustrate how the underlying mathematics can be used to find basic engine specifications. Input values are estimates.~~

~~Examples:~~

~~1. A 2 cylinder, 2-stroke, 45 hp engine.~~

~~Let: **hp** = 20; **k** =2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **F** = 1100 lbf; **Ps** = 4.5 in;~~

~~**Fw** = 6 lb/gal; **Qc** = 20500; **Lo** = .35~~

~~**Vp = 10 ft/sec** = 550(20)(2)/(2)(1100)~~

~~**bore = 1.561 in**. = 2(1100/575π)$^{.5}$ Set bore size at most frequently used 20 hp.~~

~~**Rv = 191 rpm**. = 30(10)/(.5π) Reduction gear may be required.~~

~~**Dp = 17.2 cu.in**. = π(1.561/2)^2(4.5)(2)~~

~~**Fr = .001061074 lbm/sec**. = 550(20)/(778)(20500)(1-.35)~~

~~**Fg = .6367 gals/hr**. = .001061074(3600)/6~~

~~**T = 550 lbf-ft.** = 1100(.5)~~

~~Let: **hp** = 45; **k** =2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **Vp** = 15 ft/sec.~~

~~**bore = 1.561 in**. = 2(1100/575π)$^{.5}$ bore size same as 20 hp.~~

~~**F = 1650 lbf**. = 550(45)(2)/(2)(15)~~

~~**Rv = 286 rpm**. = 30(15)/(.5π) Reduction gear probably required.~~

~~**Cp = 862 psi**. = 4(1650)/π(1.561^2)~~

~~**Fr = .002387418 lbm/sec**. = 550(45)/(778)(20500)(1-.35)~~

~~**Fg = 1.432 gals/hr**. = .002387418(3600)/6~~

~~**T = 825 lbf-ft**. = 1650(.5)~~

~~2. A 4 cylinder, 4-stroke 400 hp engine.~~

~~Let: **hp** = 50; **Vp** = 14 ft/sec; **F′** = 700 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.; **Ps** = 5.0 in;~~

~~**Fw** = 6 lb/gal; **Qc** = 20500; **Lo** = .35~~

~~**F = 1964 lbf**. = 4(50)(550)/4(14)~~

~~**Rv = 178 rpm** = 30(14)/(.75π)~~

~~**bore = 1.890 in**. = 2(1964/700π)$^{.5}$ Set bore size at most frequently used 50 hp.~~

~~**Dp = 56 cu.in**. = π(1.890/2)^{2}(5.0)(4)~~

~~**Fr = .002652686 lbm/sec**. = 550(50)/(778)(20500)(1-.35)~~

~~**Fg = 1.592 gals/hr**. = .002652686(3600)/6~~

~~**T = T′ = 1473 lbf-ft**. = 1964(.75)~~

~~Let: **hp** = 400; **Vp** = 27 ft/sec; **F′** = 800 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.~~

~~**F = 8148 lbf**. = 4(400)(550)/4(27)~~

~~**Rv = 344 rpm** = 30(27)/(.75π)~~

~~**bore = 1.890 in**. bore size same as 50 hp.~~

~~**Cp = 2904 psi.** = 4(8148)/π(1.890^{2})~~

~~**Fr = .02122149 lbm/sec**. = 550(400)/(778)(20500)(1-.35)~~

~~**Fg = 12.733 gals/hr**. = . 02122149(3600)/6~~

~~**T = T′ = 6111 lbf-ft**. = 8148(.75)~~

~~* If this engine were a 2-Stroke, there could be 50% power stroke overlap with both pairs active.~~

~~At low power, a pair of pistons could be stopped without load on the engine.~~

~~3. A 6 cylinder, 2-stroke 1200 hp. engine.~~

~~Let: **hp** = 700; **Vp** = 25 ft/sec; **F′** = 800 psi; **n** = 6; **k** = 2; **r′** = .75 ft = 9 in; **Ps** = 6.0 in.~~

~~**F = 5133 lbf**. = 2(700)(550)/6(25)~~

~~**Rv = 318 rpm** = 30(25)/(.75π)~~

~~**bore = 2.858 in**. = 2(5133/800π)$^{.5}$ Set bore size to most frequently used power.~~

~~**Dp = 231 cu.in**. = π(2.858/2)^{2}(6.0)(6)~~

~~**T = 3850 lbf-ft**. = 5133(.75)~~

~~**T′ = 1150 lbf-ft**. = 6(3850)/(2)~~

~~Let: **hp** = 1200; **Vp** = 35 ft/sec; **n** = 6; **k** = 2; **r′** = .75 ft. = 9 in.~~

~~**F = 6285 lbf.** = 2(1200)(550)/[(6)(35)]~~

~~**Rv = 446 rpm** = 30(35)/(.75π)~~

~~**bore = 2.858 in**. bore size same as 700 hp.~~

~~**Cp = 980 psi.** = 4(6285)/π(2.858^{2})~~

~~**T = 4714 lbf-ft.** = 6285(.75)~~

~~**T′ = 1414 lbf-ft**. = 6(4714)/2~~

~~4. An 8 cylinder (2 banks of 4 cyls. each), 4-stroke, 1200 hp engine.~~

~~Let: **hp** = 1200; **F′** = 800 psi; **n** = 8; **k** = 4; **Vp** = 35 ft/sec; **r′** = 1.25 ft; **Ps** = 6.0 in.~~

~~1 cyl. per 1-way clutch requiring eight 1-way clutches. 50% overlap.~~

~~**Rv = 267 rpm** = 30(35)/(1.25π)~~

~~**F = 9429 lbf**. = 4(550)(1200)/[(8)(35)]~~

~~**bore = 3.874 in**. = 2(9429/800π)$^{.5}$~~

~~**Dp = 566 cu.in**. = π(3.874/2)^{2}(6)(8)~~

~~**T = 11786 lbf-ft**. = 9429(1.25)~~

~~**T′ = 23573 lbf-ft**. = 8(11786)/4~~

~~5. A large 8 cylinder, 2-stroke, 8,000 hp marine engine.~~

~~Let: **hp** = 8000; **F′** = 800 psi; **n** = 8; **k** = 2; **Vp** = 28 ft/sec; **Rv** = 100 rpm; **Ps** = 10 in; **Lo** = .35;~~

~~**Fw** = 7.1 lbm/gal. (1 cyl./1-way clutch uses 8 1-way clutches. 75% power stroke overlap.)~~

~~**F = 39286 lbf.** = 2(550)(8000)/[(8)(28)]~~

~~**r′ = 2.673 ft**. = 30(28)/(100π) Units 89 (FIGs 7,8) are carried by a short rimmed outer race 5~~

~~with a single spoke 35 to reduce inertia.~~

~~**bore = 7.907 in**. = 2[(39286/800π)]$^{.5}$ Set bore size to most frequently used power.~~

~~**Dp = 3929 cu.in**. = π(7.907/2)^{2}(10)(8)~~

~~**Fr = .457600229 lbm/sec.** = 550(8000)/(778)(19014)(1-.35)~~

~~**Fg = 232.02 gals/hr**. = .457600229(3600)/7.1~~

~~**T = 105011 lbf-ft**. = 39286(2.673)~~

~~**T′ = 420046 lbf-ft**. = 8(105011)/2~~

~~***** Reset power by activating/deactivating piston pairs then vary power by varying the fuel charge.~~

Interchanging 4-Stroke and 2-Stroke.

With reference first to FIG 3, the flexible chain 9 must be made stiff enough to pull the piston 38 down during the intake stroke in the 4-stroke engine. In this case, the outer race 5 is a cogwheel and the cogs fit between the chain links similar to a bicycle chain. Each side of each link has an extension that rides in an immovable channel that is secured to the engine. The two channels combine with the side extensions to prevent the chain from flexing out of mesh with the race 5 cogwheel during the intake stroke and without interfering with the other piston strokes. Both channels are shaped in an arc around the race 5 where they are connected with a solid cover over the chain to insure against the chain flexing. The cover ends near the position of fastener 41 in FIG 3 when piston 38 is at top dead center. The channels continue straight downward without the cover to prevent the chain from flexing when it is straight. The straight channels extend to a point slightly beyond the position of fastener 41 when the piston is at bottom dead center. The fastener is connected to the chain free of the extensions so that the channels do not interfere with the motion of the chain. This allows the fastener 41 to reach its highest point (FIG 3) where it is in position to begin the intake stroke and complete the stroke without interference from the channels or the cover.

~~There are at least two simple ways to effect this change.~~ There are at least two simple ways to change between a 2-stroke and a 4-stroke. In a 4-stroke, a sector gear 12 on two pairs engages idler 40A (FIG 6). A removable cap 54 having a hole is ~~removably secured, e.g.~~ threaded[[,]] to the engine 15. The shaft 43 of idler 40A has two diameters. The shorter one extends through the hole. A snap ring 56 on the shorter diameter abuts the cap and combines with the larger diameter that abuts the inside of the cap to prevent the idler 40A from axial movement which keeps the idler properly engaged with the two sector gears. When changing to a 4-stroke from a 2-stroke, the pistons must be correctly ~~aligned~~ positioned before engaging the idler with the sector gears. One of the correct ~~alignments~~ positions is shown in FIG 6 with 2 pistons at top dead center and 2 at bottom dead center. Power stroke overlap for a 4-stroke can be achieved by adding another bank of two pairs along the shaft 8 disengaged from the bank shown in FIG 6 or by adding separate pairs. ~~To avoid cluttering, FIGs 6,6A show the splined end of shafts 43 without flywheels 48.~~

The separation 1 in FIG 6A makes the 4-stroke a 2-stroke. To change to a 2-stroke from a 4-stroke, the cap 54 is partly unscrewed to a predetermined position on the engine 15, which raises shaft 43 and disengages idler 40A from sector gears 12 ~~(FIG 8A)~~ (FIG 6A). The cap is held in place by known means, e.g. a dowel through the side of the cap that contacts engine 15.

~~Alternatively, for a 4-stroke, one or more dowels through engine 15 engage a radial groove in shaft 43 to prevent axial movement but allows rotary motion of idler 40A while engaging sector gears 12. To change to a 2-stroke, the dowels are removed from the groove. Idler 40A is separated~~

~~from sector gears 12 by lifting shaft 43 to where the dowels are inserted in a second groove. Shaft 43 is lifted to where the dowels are inserted in a second groove, which separates idler 40A from sector gear 12.~~

The second mechanical version is shown in FIG 12. Some reference numbers for the same parts in FIG 11 are omitted in FIG 12 to avoid overcrowding. ~~In FIG 12, the rod 101 is discarded by connecting one arm of the lever 100 directly to the wrist pin 97. A slant 25 of the contact surfaces is provided between the piston 81 and race 4. The spring 11 in FIG 13 can be included.~~ A lever 100 oscillates on its fulcrum 99 which extends from race 4. A gear mesh combines lever 100 with rod 84 to shift piston 81 into and out of contact with surface 112 on race 5. The piston is shown in contact with surface 112. The single piece rod 84 and piston 81 shift along a clutch radial 93 while in sliding contact with the carrying race 4. Space 88 allows the shift. Only a few teeth complete the gear mesh since the rod's motion is very short. A very short motion reduces backlash and may even make it negligible. If short enough, the gear mesh could be eliminated in favor of a single piece lever and rod. The spring-loaded trigger 85 at the end of arm 3 extends across gap 28 and stays in contact with the tough, long wearing strip 14 carried by race 5. The piston never contacts the strip 14. The trigger slides over strip 14 during overrun and grabs it at the beginning of the power stroke to oscillate the lever in response to the motion of race 5, thereby shifting the rod and piston. Torque is thus efficiently transmitted to race 4 perpendicular to the clutch radial 93.

Not shown is a third mechanical version that sets the piston on one radial of the clutch and the fulcrum on another. It can also eliminate the rod 101.

In all the 1-way clutch embodiments shown in FIGs 9-11,13: (1) the angle at the trigger's two extreme positions must not cause jamming, (2) the trigger should be suitably coated ~~with a suitable ceramic~~ and shaped to reduce drag but instantly grab the outer race when reversing to the drive direction, (3) the piston's motion 88 goes only far enough to provide clearance between the piston and the outer race during overrun and (4) one of the contact surfaces has a common V-groove and the other contact surface is beveled to fit it to prevent slip.